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EXPERIMENTAL INVESTIGATION OF A STEAM-WATER INJECTOR WITH A TAPERED NOZZLE

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Working recommendations for determining the basic parameters and geometry of a steam–water injector with a converging nozzle are presented on the basis of experimental data. Injector startup, operating efficiency, and service life are examined.

Investigations of an injector with a tapered nozzle were performed at the Russian Science Center Kurchatov Institute for the purpose of developing a circulator for the first loop of a nuclear power system. A planar model of a steam-water injector with two transparent walls, used to show the real picture of the processes in the flow-through part, was investigated at the first stage. Next, experimental models of an injector with mixed nozzles and mixing chambers were studied. This made it possible to develop working samples meeting the prescribed conditions.

Injector models with a mixing chamber were investigated with taper angles 3° and 12°, diffusor throat diameters 2.75, 3, and 3.2 mm, and nine values of the exit diameter of the tapered steam nozzle ranging from 3.8 to 5.4 mm. The combination of changeable mixing chambers and steam nozzles made it possible to test 14 injector models. The position of the steam nozzle relative to the entrance into the mixing chamber could be changed by using a moving apparatus during the experiment, i.e., the flow-through section of the water nozzle could be changed. The investigations resulted in the development of three working samples of an injector with mixing chamber taper angle 12°, diffusor throat diameter 3.1 mm, and exit diameter of the tapered steam nozzle 4 mm. The samples differed only by the water flow-through sections.

More than 1000 regimes of working injectors were selected and tested. A working injector is an apparatus with exit pressure higher than the entrance pressure. As a result, certain working recommendations were obtained for injectors operating at steam pressure below 4 MPa.

Steam Flow Rate. The steam flow rate through a working nozzle is determined in practice from the well-known equation

$$G_{st} = k_n f_n \sqrt{p_1 / v_1},$$

where p_1, v_1, and f_n are, respectively, the pressure and specific volume of the steam at the entrance, and the flow-through section of the nozzle (Fig. 1*a*); and, the factor k_n depends on the adiabatic exponent.

Injector models with conical steam nozzles were investigated. The profile of the nozzles of the working samples was calculated using the Vitoshinskii formula. For the first set $k_n = 1.77$ and for the second set $k_n = 1.84$. The computational accuracy for the steam flow rate of a Vitoshinskii nozzle is 3.5–5%. The counterpressure of the injector has an appreciable effect on the flow rate of the steam, and consequently its value was set for the maximum load.

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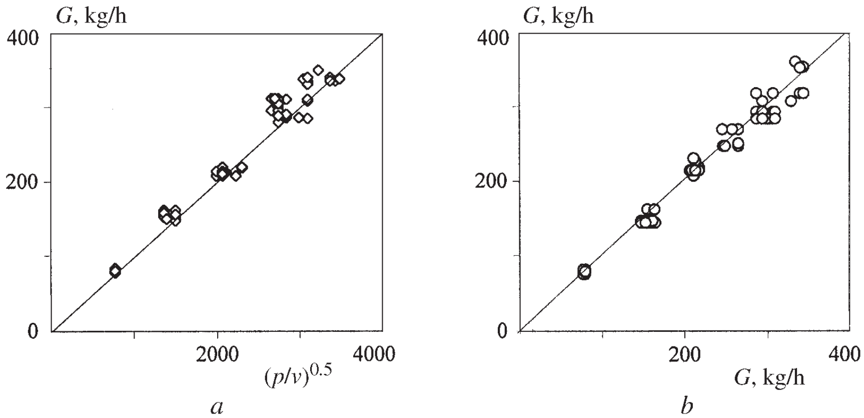

Fig. 1. Experimental dependence of the steam flow rate on $(p/v)^{0.5}$ (*a*), comparison of experimental and computed steam flow rate (*b*).

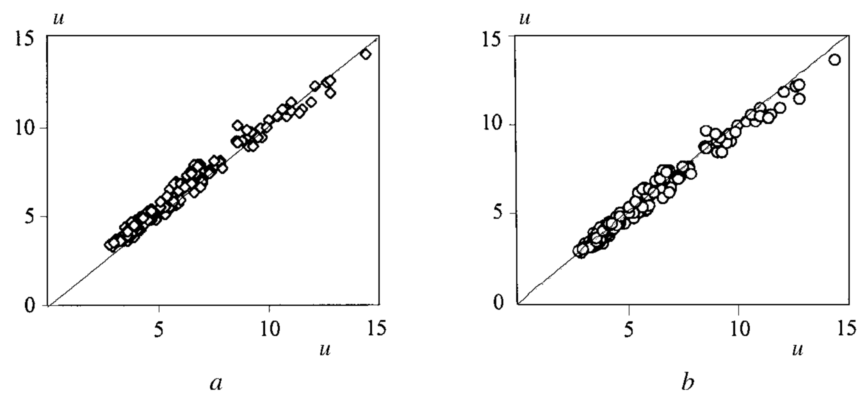

Fig. 2. Comparison of the computed and experimental injection coefficient: *a*, *b*) calculation using Eqs. (1) and (2), respectively.

Injection Coefficient. For injection coefficient above 8, the best agreement between the calculations and the experimental data (4.5–5.5%, Fig. 2*a*) is obtained with the equation

$$u = 1/\ln\left(\frac{i_1 + i_{\text{in}}}{i_1 + i_{\text{out}}}\right),$$

and for injection coefficient below 8 (Fig. 2*b*)

$$u = \frac{i_1 + i_{\text{out}}}{i_1 + i_{\text{in}}},$$

where i_1 is the steam enthalpy at the nozzle entrance.

The calculations of the injection coefficient based on the heat balance in the mixing chamber gives an up to 20% disagreement with experiment.

Water Nozzle. One problem considered in this investigation was determining the flow through section of the water nozzle or the entrance of injected water into the mixing chamber of the injector. The difficulty in calculating the section of

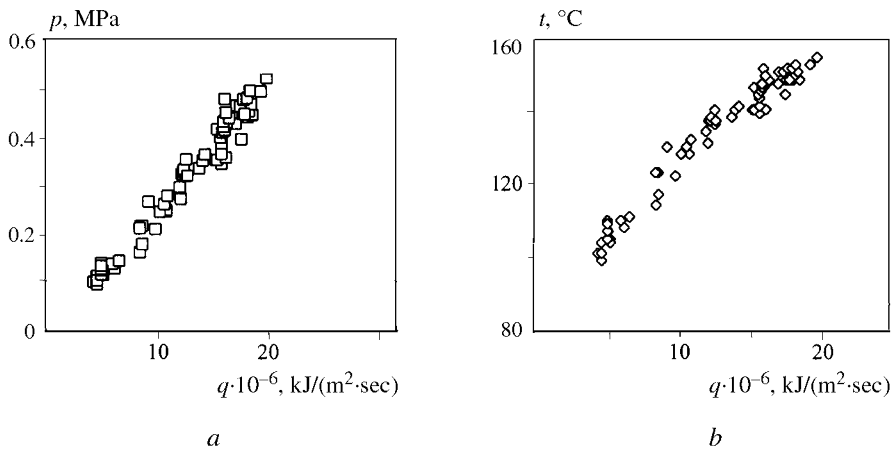

Fig. 3. Pressure (*a*) and temperature (*b*) in the mixing chamber versus the relative thermal power of the injector.

the water nozzle lies in the uncertainty of the pressure in the mixing chamber. Measurements of local parameters in the flow through part of transparent models of an injector shed little light on anything; the results of the investigation are more qualitative than of quantitative value. The pressure on the wall of the mixing chamber decreases from the entrance value to a value that can be higher or lower than the saturation pressure corresponding to the temperature of the flow at the exit. The pressure on the edge of the steam nozzle at the water–steam boundary is lower than on the wall and equalizes downstream. Measurements of the local parameters only in three sections of the mixing chamber made it impossible to judge the average pressure there. Consequently, ordinarily it is necessary to examine various versions of the calculation of the pressure in the mixing chamber.

The most common version is determining the average pressure in the mixing chamber according to the saturation pressure, which corresponds to the temperature at the injector exit $p_{\text{m.c}} = p_s(t_{\text{out}})$. For the injectors investigated, it is not suitable for at least two reasons: in a substantial number of the regimes $p_s(t_{\text{out}})$ of the water is higher than the pressure at the entrance, and the computed entrance resistance coefficient of the injected water is a factor of 10 or more smaller than the experimental value.

Analysis of the experimental data showed that the pressure in the mixing chamber depends on all parameters and the geometry of the injector:

$$p_{\text{m.c}} = f(t_{\text{in.w}}, t_{\text{out.w}}, p_{\text{st}}, u, f_{\text{w}}, f_{\text{st}}).$$

The simplest method of determining the average pressure in the mixing chamber is on the basis of the difference of the experimental water pressure at the entrance and the computed pressure difference at the entrance. The independent variable was chosen to be a complex containing many of the parameters enumerated above:

$$q = D_{\text{st}}(1 + u)i_{\text{out}}/f_{\text{st}}.$$

This is essentially the thermal power scaled to the area of the steam exit section.

The computed pressure in the mixing chamber can be represented as a linear function (Fig. 3*a*)

$$p_{\text{m.c}} = k_p q,$$

and the temperature corresponding to this pressure can be represented by a power-law (Fig. 3*b*)

$$t_{\text{m.c}} = k_t q^a,$$

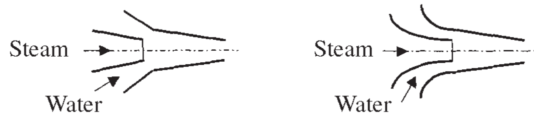

Fig. 4. Sketches of the construction of the water entrance into the mixing chamber of the injector.

where the coefficients k_p and k_t take account of the relative geometric dimensions of the water entrance into the mixing chamber of the injector and are represented as a quadratic trinomial

$$k_{p,t} = a(f_w/f_{st})^2 - b(f_w/f_{st}) + c. \tag{1}$$

The injectors were investigated for values of the ratio f_w/f_{st} = 0.6–4 with equivalent diameter of the water nozzle d_{eq} = 0.9–5.7 mm.

For working samples of the injector, the entrance of the injected water was coaxial to the steam flow, whereas for the models water enters at an angle. This unquestionably influenced the compression of the flow. In addition, the variable flow-through section was reflected on the form of the initial section of the mixing chamber (Fig. 4), and consequently the coefficients in Eq. (1) for the models are different from those for the working samples. The investigations of the injectors were of an exploratory nature, for purposes of developing a better construction. The construction influenced not only the entrance but also the cutoff pressure. Consequently, the q dependences of $p_{m.c}$ and $t_{m.c}$ are presented for the most improved constructions of the injectors investigated:

$$p_{m.c} = (-2.63(f_w/f_{st})^2 + 7.84(f_w/f_{st}) - 4.59)10^{-6}q; \tag{2}$$

$$t_{m.c} = (-69.3(f_w/f_{st})^2 + 209.2(f_w/f_{st}) - 52.45)10^{-6}q^{0.26}. \tag{3}$$

The entrance pressure of the water, calculated using the relations (2) and (3), agrees with the experimental value with an average error of 4.5–6%.

The assumption that the injected water flow is compressed on entering the mixing chamber is confirmed by visual observations of the transparent models of the injector and by measurements of the pressure at the nozzle edge. Consequently, the introduction of the water flow rate coefficient at the entrance into the mixing chamber into the calculation could be justified.

Analysis of the experimental data showed that the ratio of the entrance momenta of the water and steam can be taken as an independent variable flow rate coefficient. The fact that the water and steam momenta can be calculated only on the basis of a particular version makes it more difficult to solve the problem.

To determine the counterpressure of the water nozzle or the average pressure in the mixing chamber, a different extreme case, in addition to the rejected version $p_{m.c} = p_s(t_{out})$, was also examined. The counterpressure of the water nozzle is determined by the average temperature of the injected water, which varies from t_{in} to t_{out}, provided that t_{out} does not exceed $t_s(p_w)$. If $t_{out} > t_s(p_w)$, then $t_{out} = t_s(p_w)$:

$$p_{m.c} = p_s[0.5(t_{in} + t_{out})],$$

where p_s is the saturation pressure for the average value of the entrance and exit temperature of the injected water and p_w is the experimental value of the water pressure at the entrance.

According to this version the average pressure in the mixing chamber can be determined more accurately using the formula

$$P_{m.c} = \sum_{t=t_{in}}^{t=t_{out}} p_s \Delta t(t_{av} + t_{in}), \tag{4}$$

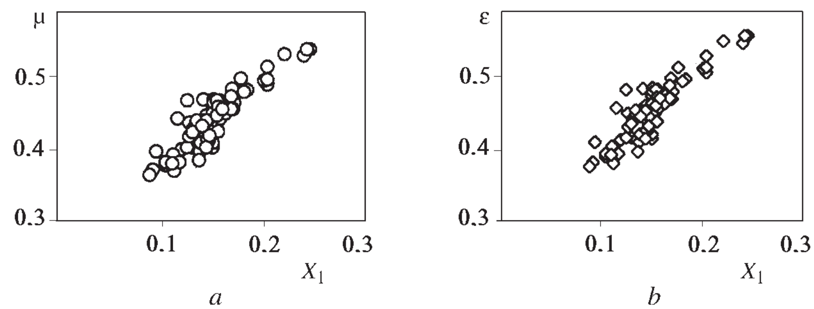

Fig. 5. Experimental dependence of the order flow rate coefficient (*a*) and the flow compression coefficient (*b*) on the ratio of the water and steam momenta.

TABLE 1. Coefficients in Eqs. (6) and (7) for Calculating $p_{m.c}$ Using Eq. (4)

Temperature t_2 at the exit of the steam nozzle	*a*	*b*	*c*	*d*	δ, %
$t_2 = t_{in}$	0.4	0.036	0.256	1.316	4.9–5.0
$0.5(t_{in} + t_{out})$	0.4	0.994	3.372	3.935	4.4–6.0
$0.5(t_{cr} + t_{out})$	0.4	0.843	2.872	3.332	4.1–4.8
$t_2 = t_{m.c}$	0.4	0.688	2.384	3.12	4.1–5.7
$t_2 = t_{cr}$	0.23	0.615	2.046	2.412	2.6–3.2

where p_s is the saturation pressure for the instantaneous temperature of the injected water from t_{in} to t_{out} under the same condition that t_{out} does not exceed $t_s(p_w)$. Of course, because of the thermal resistance of the phase transition the pressure in the mixing chamber should be higher than the computed value.

The ratio of the entrance momenta of the water and steam was calculated in the form

$$X_1 = \frac{u\varphi_2\sqrt{2g(p_w - P_{m.c})} + p_w f_w}{\varphi_1\sqrt{(2g\Delta_i / A)} + p_{st} f_{st}}, \tag{5}$$

where u is the injection coefficient; $\varphi_1 = 0.96$ is the velocity coefficient of the steam nozzle; φ_2 is the computed velocity coefficient of the water nozzle; Δ_i is the heat differential in the steam nozzle; p_{st}, p_w, and $p_{m.c}$ are, respectively, the steam pressure and the water pressure at the entrance, and the pressure in the mixing chamber; f_{st} and f_w are the exit section of the steam and water.

To compute the entrance momentum of the steam, the temperature at the exit of the steam nozzle was taken as follows: $t_2 = t_{out}$, $t_2 = 0.5(t_{in} + t_{out})$, $t_2 = 0.5(t_{cr} + t_{out})$, $t_2 = t_{m.c}$, $t_2 = t_{cr}$.

To eliminate any influence of the resistance coefficient of the entrance into the mixing chamber, the compression coefficient of the injected flow was studied, instead of the flow rate coefficient, as a function of the ratio of the input momenta of the water and steam. As an example, Fig. 5*a* shows the experimental dependence of the flow rate coefficient and the compression coefficient of the injected flow (Fig. 5*b*) as a function of the ratio of the water and steam momenta for $t_{m.c} = 0.5(t_{in} + t_{out})$ and $t_2 = t_{m.c}$, which is described by a power-law function of the type

$$\varepsilon = k_\varepsilon X_1^a, \tag{6}$$

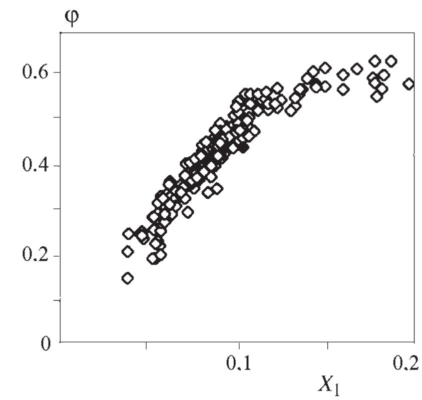

Fig. 6. Experimental dependence of the coefficient $\varphi^2_{\text{m.c+dif}}$ on the momentum ratio.

where the coefficient k_ε takes account of the geometry of the water entrance in the mixing chamber and is written as a quadratic trinomial

$$k_\varepsilon = b(f_{\text{w}}/f_{\text{st}})^2 - c(f_{\text{w}}/f_{\text{st}}) + d. \tag{7}$$

The constant coefficients *a*, *b*, *c*, and *d* in Eqs. (6) and (7) are presented in Table 1.

The relations presented above for calculating the pressure in the mixing chamber are approximately equivalent. How the momentum of the steam is determined and its actual value are not important. The parameters used are related with one another by constants obtained on the basis of the experimental data.

Mixing Chamber. It is well known that the main losses in the mixing chamber are shock losses, which are unavoidable for mixing of flows with different mass and velocity. To determine the efficiency of the mixing chamber and the diffusor it is necessary to know the entrance momentum of the steam and water. Investigations of transparent models showed that the steam at the exit from a tapered nozzle in the mixing chamber expands along the center approximately up to the temperature at the entrance and along the periphery up to an unknown temperature from t_{in} to t_{cr} or some intermediate value.

Of the five possible computational variants of the temperature t_2 at the exit of the steam nozzle: $t_2 = t_{\text{out}}$, $t_2 = 0.5(t_{\text{in}} + t_{\text{out}})$, $t_2 = 0.5(t_{\text{cr}} + t_{\text{out}})$, $t_2 = t_{\text{m.c}}$, $t_2 = t_{\text{cr}}$ – the fourth variant is studied in order to simplify the computational procedure; the fifth variant is unsuitable because of the high velocity coefficient of the mixing chamber and diffusor $\varphi_{\text{m.c+dif}}$. Its value was close to and even greater than 1 in many of the experimental regimes.

Going against convention, the energy losses in the mixing chamber and diffusor were studied together. These losses contain everything possible, including an incorrect choice of parameters in the flow through part, except for shock losses.

Figure 6 shows the experimental dependence of the velocity coefficient of the mixing chamber and diffusor $\varphi^2_{\text{m.c+dif}}$ on the ratio of the momenta according to Eq. (5). The choice of the ratio of the water and steam momenta as the independent variable is made not only to unify the computational procedure but also because nothing better was found. The computed dependence of the velocity coefficient of the mixing chamber and diffusor on the momentum ratio is given by the equation

$$\varphi_{\text{m.c+dif}} = \sqrt{a + k_{\text{m.c+dif}} X_1}\,, \tag{8}$$

where the coefficient $k_{\text{m.c+dif}}$ takes account of the ratio $f_{\text{w}}/f_{\text{st}}$ of the water and steam entrance section. It is represented as the quadratic trinomial

$$k_{\text{m.c+dif}} = c(f_{\text{w}} / f_{\text{st}})^2 + d(f_{\text{w}} / f_{\text{st}}) + e. \tag{9}$$

The constant coefficients *a*, *b*, *c*, *d*, and *e* in Eqs. (8) and (9) are presented in Table 2 as a function of the temperature at the exit of the water and steam nozzles.

TABLE 2. Coefficients in Eqs. (8) and (9) for Calculating $p_{m.c}$ Using Eq. (2)

Exit temperature t_2 of the steam nozzle	a	b	c	d	e	φ^2_{max}	δ, %
$t_2 = t_{in}$	–0.6	0.33	–0.576	2..07	–0.104	0.35	7.3–8.1
$0.5(t_{in} + t_{out})$	–0.6	0.25	–0.223	0.830	0.622	0.26	5.4–6.3
$0.5(t_{cr} + t_{out})$	–0.6	0.50	–1.40	4.916	–1.82	0.50	6.8–7.4
$t_2 = t_{m.c}$	–0.6	0.25	–0.133	0.540	0.867	0.27	6.1–7.7

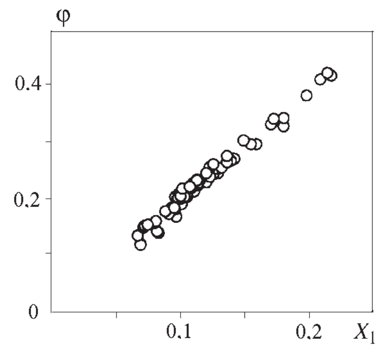

Fig. 7. Experimentally determined volume steam content in the throat of the injector diffusor versus the momentum ratio.

When X_1 reaches a certain value, for most of the investigated injectors the value of $\varphi^2_{m.c+dif}$ actually no longer changes; the tables give the limiting value of this quantity φ^2_{max}. Table 2 gives the computed error of the edge pressure δ.

Diffusor. The flow-through section of the diffusor throat is determined according to the volume steam content and the velocity of the mixed flow. The difficulty of the problem lies in the fact that these two parameters are unknown.

The volume steam content in the diffusor throat can be expressed as the ratio

$$\varphi_t^{cal} = w_t / w_{cal},$$

where

$$w_t = (G_{st} + G_w) v_w / f_t;$$

$$w_{cal} = \frac{\sqrt{2g/A} + \sqrt{2gv_w(p_w + p_{m.c})}}{1+u},$$

w_t is the average velocity in the diffusor throat with complete condensation of the steam in the mixing chamber; G_{st} and G_w are, respectively, the steam and water flow rates; f_t is the flow through section of the diffusor throat; w_{cal} is the velocity computed according to the momentum equation neglecting slipping and without the coefficient $\varphi_{m.c+dif}$; and, for a more accurate form of the momentum equations, the quantities $f_{st}p_{st}$, $f_w p_w$, and $f_t p_t$ had no effect on computational accuracy.

Figure 7 shows the experimentally determined volume steam content in the throat of the injector diffusor as a function of the ratio of the momenta calculated according to Eq. (5) with $t_2 = t_{out}$ and $t_{m.c}$.

The computed volume steam content in the diffusor throat was represented, on the basis of an analysis of the experimental data, by the equation

$$\varphi_t = k_\varphi X_1^a, \tag{10}$$

TABLE 3. Coefficients in Eqs. (10) and (11) for Calculating $p_{m.c}$ Using Eq. (2)

Temperature t_2 at the exit of the steam nozzle	a	b	c	d	δ, %
$t_2 = t_{in}$	1	0.027	0.098	0.47	3.3–5.0
$0.5(t_{in} + t_{out})$	1	–0.08	1.26	–2.78	3.4–5.2
$0.5(t_{cr} + t_{out})$	1	0.145	–1.21	3.99	3.5–5.2
$t_2 = t_{m.c}$	1	–0.07	1.18	–2.56	3.3–5.1

where

$$k_\varphi = b + c(f_{st} + f_w)/f_t, \tag{11}$$

the coefficient k_φ takes account of the injector geometry, f_{st} is the exit section of the steam nozzle, f_w is the exit section of the water nozzle, and f_t is the flow through section of the diffusor throat.

The constant coefficients a, b, c, and d in Eqs. (10) and (11) are given in Table 3 as a function of the temperature at the exit of the water and steam nozzles. The computed dependences are far from the real volume steam content in the diffusor throat, but they can be used for calculations.

The length of the diffusor throat has no effect on the operation of the injector. For working samples of the injector it is ~5 mm and was determined by technological factors. Preliminary investigations showed that the flare angle of the diffusor is of no great importance, and consequently the diffusor was calculated with the same flare angle 12°.

On the basis of the results of the investigations of injectors it can only be asserted that the velocity and temperature of the injected water have an appreciable influence on the length of the mixing chamber. Above the optimal length of the mixing chamber the steam condenses at a shorter distance from the nozzle cutoff and some of the energy of the mixed flow is lost on deceleration and acceleration of the flow. This was observed in some transparent models and injectors with a 3°taper angle of the mixing chamber. For the working injector samples the taper angle of the mixing chamber is 12°. Investigations were not performed with a larger angle. Consequently, it can only be asserted that a taper angle of less than 12° is not recommended.

Structural Features of the Injector. The construction of the investigated injectors and the flat transparent models is based on a scheme with a central feed for steam and peripheral feed for water; the injected water first enters a receiving chamber and then the mixing chamber. This scheme is conventional and has drawbacks which degrade the operation of the injector – heat transfer from steam to water in the receiving chamber before they mix in the mixing chamber and existence of stagnation zone in the receiving chamber. A decrease of the heat content of steam decreases the momentum of the steam, just as an increase of the heat content of water, as a result of an increase in pressure at the exit of the steam nozzle. The presence of heated water in the stagnation zone complicates the startup process and can make it impossible. The difference between the temperature at the entrance into the injector and at the location where the water enters the mixing chamber reached 30°C in flat models.

Changes in the injector structure were made not only to change the geometry of the flow-through part but also to decrease the heat transfer from steam to water. Although the steam and water could not be completely isolated, for technological reasons, the surface where steam and water touched in the working samples of the injector was decreased in size, which affected the characteristics of the injectors.

A working injector characteristically emits noise, a special "voice," according to which experimenters distinguish a working injector. The collapse of steam bubbles in the condensation jump in the expanding flow through part makes it ring like a bell. The noise level and frequency were determined by placing the condensation jump in the diffusor. As counterpressure increased, and the jump shifts to the diffusor throat, the frequency increases and vice versa. An external water jacket of the receiving chamber decreases the noise level of the working samples.

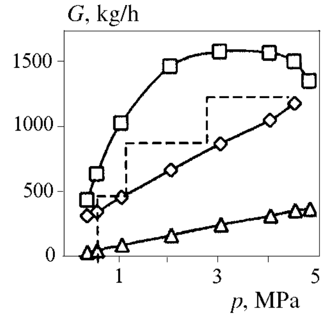

Fig. 8. Steam flow rate (△) and the minimum (◇) and maximum (○) water flow rates versus the steam pressure at the injector entrance, possible change in water flow rate and steam pressure with forced startup of the injector (– – –).

Injector Startup, Efficiency, Service Life. The detractors of injectors consider startup together with low efficiency as the main drawbacks. Indeed, an investigator encountering an injector for the first time will have difficulties in starting it. As a rule, difficulties in starting an injector arise during discrete operation, for example, in a steam engine, where it is used as a makeup pump, and in experimental investigations. There are several ways to facilitate startup in this case: from overlow lines and mobile nozzles to influencing the input parameters of an injector. Such startup is called forced.

The startup method using overflow lines consists in creating a counterpressure to the steam nozzle by dumping some of the coolant into an open space or tank with a lower pressure than in the loop. As a rule, the water to be dumped is extracted before the diffusor throat – the narrowest place in an injector. The overflow lines are used to start up a steam-engine injector, but most often for starting an injector in a closed loop. The main drawbacks include distortion of the profile of the mixing chamber, which is a source of additional losses. These drawbacks can be eliminated only if the requirements for the efficiency of heat conversion in the injector are low.

For central water feed into the injector, moving the water nozzle closer to the diffusor throat facilitates injector startup. Naturally, this makes the injector construction much more compicated, but a distinguishing property of an injector is lost – absence of moving parts in its structure.

Injectors at the Russian Science Center Kurchatov Institute were investigated in open and closed loops. In all stands startup was performed by changing the input parameters.

Figure 8 shows the flow rate characteristic of a working injector – the dependence of the flow rate of the injected water on the steam pressure at the entrance into the injector. The field between the top curve – maximum water flow rate – and the bottom curve – minimum flow rate – is the working space of the injector. Below the minimum flow rate of water the injector becomes steamed up, i.e., uncondensed steam appears at the exit. For water flow rate above the maximum value, the injector is flooded with water, and the mixed flow condenses before the diffusor throat. When the injector is started up its parameters must lie in the working field, as shown by the broken line. At pressures below 0.3 MPa there were difficulties resulting from the horizontal arrangement of the working sample of the injector, and the flow-through section of the water nozzle was not completely filled, which caused the parameters to oscillate during startup. When the injector operates continuously in a closed loop, there are no startup difficulties; startup is conducted together with the parameters set close to zero, if the flow rate characteristic of the injected flow does not exceed the fall outside the values for the water flow rate.

The efficiency of the best injector investigated was low – 5–13% depending on the working parameters. The term “natural circulation efficiency” and especially its value (thousandths of a percent) are difficult to find in the technical literature. But, in either case, the heat is converted into motion inside the loop, and as usual it has no effect on the efficiency of the power system. Consequently there is no sense in studying the efficiency of an injector, just as natural circulation. But it should be noted that an injector is a circulator of a higher level than natural circulation, but only because its head is much higher. The presence of supersonic flows in it precludes the appearance of hydrodynamic instability in the loop.

It is widely believed that the service life of injectors is limited. Low estimates of the service life are based on investigations of models whose scale is M 1:1000 and lower. Naturally, the relative dimensions of a model in operation change more rapidly than a natural object. Far from perfect designs are often subjected to lifetime tests, and this is done almost always under maximum loads. The maximum pressure increase of an injector is fixed by arranging the condensation jump near the diffusor throat. When steam bubbles collapse in the condensation jump, the surface can be subjected to a substantial destruction. To prevent this the injector must be underloaded, and the initial parameters must be chosen so that the injector would not operate in a limiting regime. Then the jump lies upstream in the diffusor, where the water layer is a damper for the wall of the apparatus.